

July 26, 1966

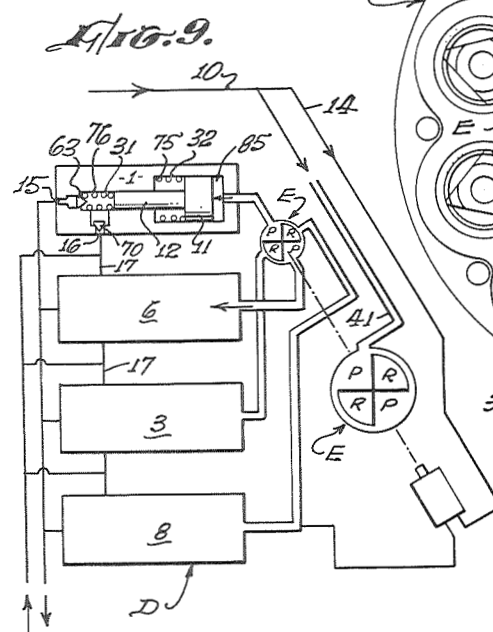
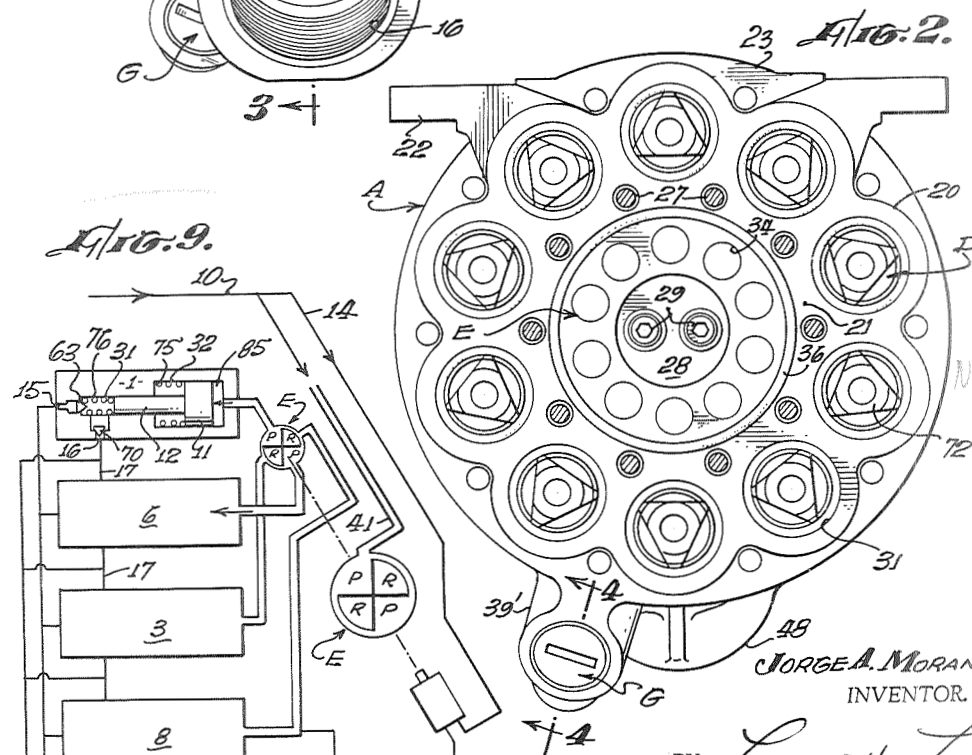
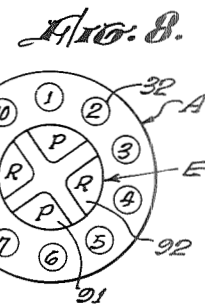
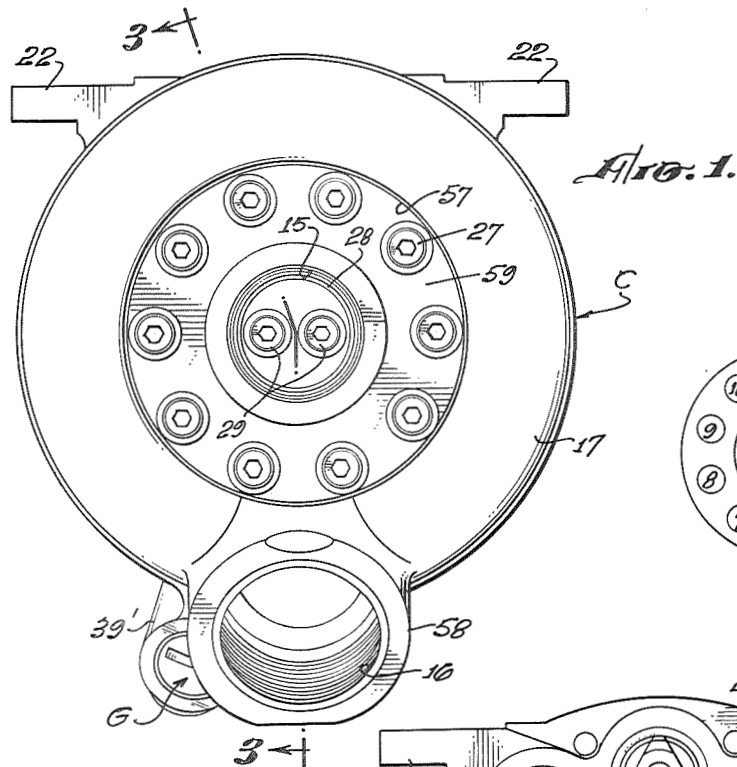
J. A. MORANDO

3,262,395

HYDRAULIC TRANSFORMER

Filed June 10, 1963

3 Sheets-Sheet 1



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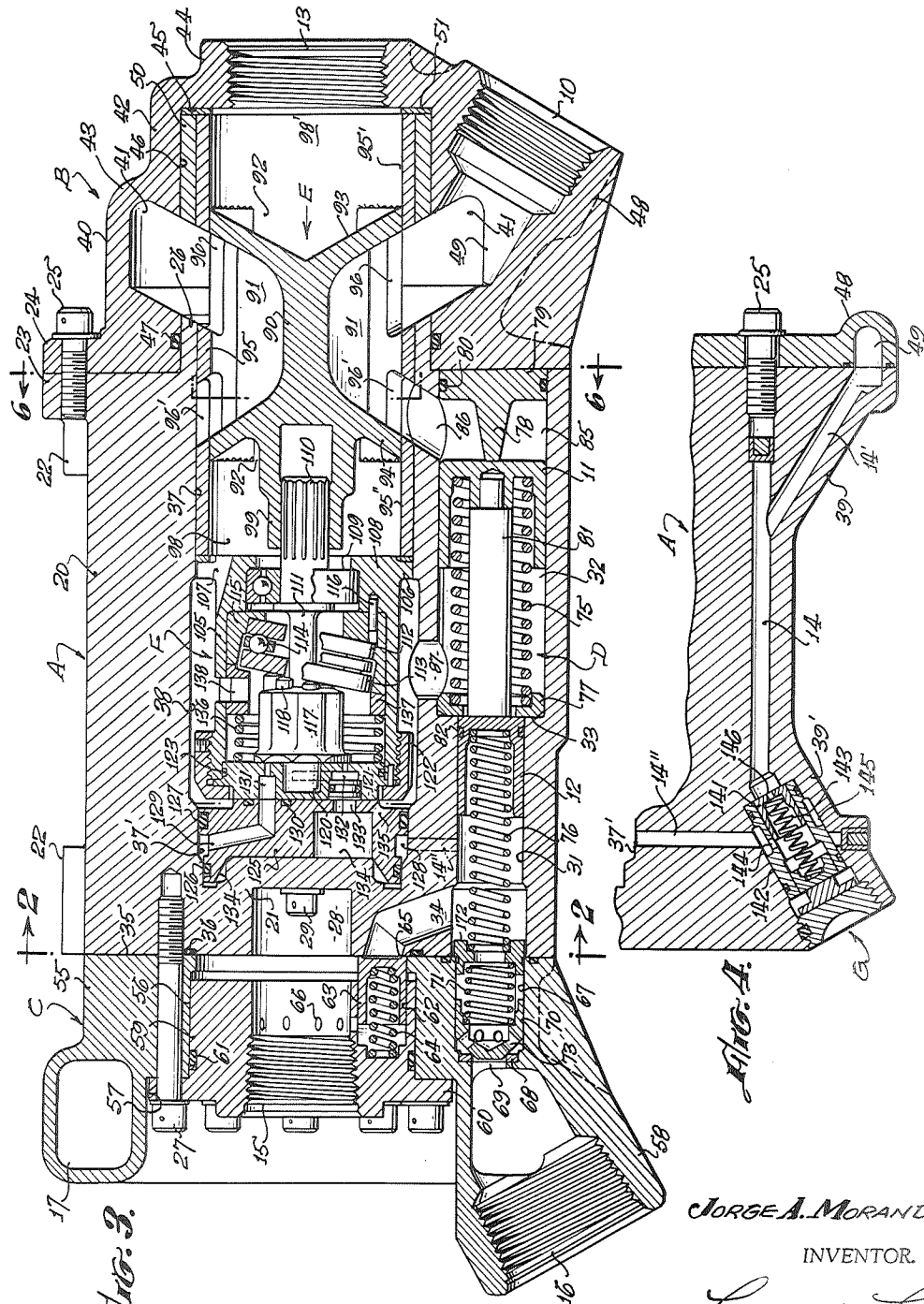
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HYDRAULIC TRANSFORMER

Filed June 10, 1963

3 Sheets-Sheet 2



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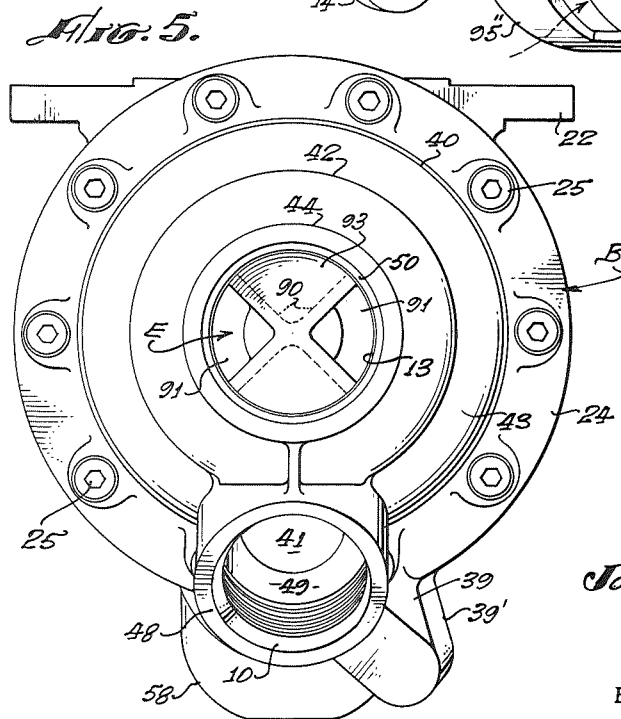
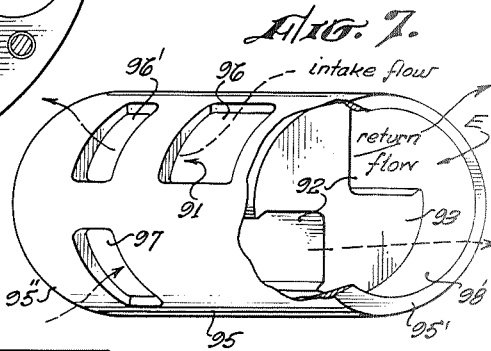
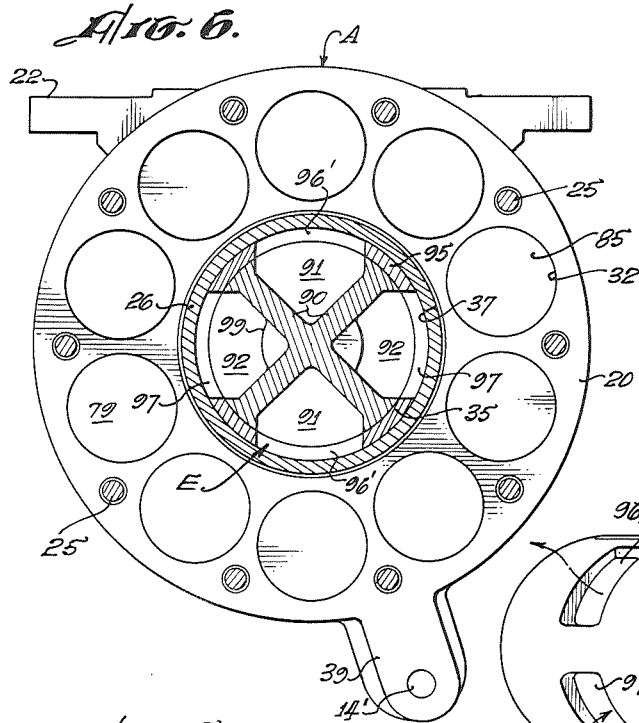
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HYDRAULIC TRANSFORMER

Filed June 10, 1963

3 Sheets-Sheet 3



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1

3,262,395

HYDRAULIC TRANSFORMER

Jorge A. Morando, Sherman Oaks, Calif., granted to National Aeronautics and Space Administration under the provisions of 42 U.S.C. 2457(d)

Filed June 10, 1963, Ser. No. 286,620

11 Claims. (Cl. 103—49)

This invention relates to variable displacement fluid pumping apparatus, and has as its principal object to provide a fluid pumping device of extremely light weight such as to adapt it especially for outer space flight vehicles wherein weight-saving is of prime consideration.

Toward the attainment of this broad object, the invention provides a fluid pressure transformer apparatus which utilizes a fluid under pressure at one level to drive a series of free pistons in a positive displacement pump which delivers a hydraulic fluid at a different pressure level to a load (such as one or more hydraulic actuators or servo motors) utilized in controlling the travel of the vehicle or other phases of operation of the environmental apparatus in which my hydraulic transformer may be utilized.

In the particular application to an outer space rocket, the invention utilizes the rocket fuel, at its normal pressure (the relatively low pressure level of the apparatus) to sequentially move a plurality of relatively large pistons of a free piston pump, and the movements of these pistons are transmitted to a corresponding series of relatively small pistons which deliver the relatively high pressure hydraulic fluid output of the transformer which is utilized to act upon the load.

In its broader aspects, the invention provides a drive fluid pumping, fluid pressure transformer system wherein a fluid driven motor drives a rotary selector control valve which directs drive fluid under pressure to a free piston positive displacement pump from which fluid at a different pressure is delivered to the load.

More specifically, the invention provides a fuel-to-hydraulic fluid pressure booster system wherein a fuel-driven hydraulic motor drives a rotary selector control valve which directs fuel under pressure to a free piston positive displacement pump from which high pressure hydraulic fluid is delivered to the load.

By utilizing a free piston pump, it is possible to avoid the necessity for using conventional rotating cylinder blocks, heavy thrust bearings and shaft seals; normal problems of efficiency, weight, size and complexity are essentially eliminated; and an extremely efficient, simple and lightweight construction is achieved.

While the invention was initially designed for use as a pressure booster or intensifier, in its broader aspects it is also applicable to a pressure de-intensifier and flow booster in which pistons of smaller diameter are acted upon by drive fluid to drive respective pistons of larger diameter which generate lower pressure and higher flow in the pump fluid than that of the driving fluid.

Other objects and advantages will become apparent in the ensuing specification and appended drawing in which:

FIG. 1 is an end view of the output end of the transformer;

FIG. 2 is an end view of the same end of the transformer with the end cap removed, and with the cap screws shown in cross-section as indicated by line 2—2 of FIG. 3;

FIG. 3 is an axial sectional view of the transformer, taken on the line 3—3 of FIG. 1;

FIG. 4 is a longitudinal sectional view taken on the line 4—4 of FIG. 2;

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FIG. 5 is an end view of the inlet end of the transformer;

FIG. 6 is a cross sectional view of said inlet end, taken on the line 6—6 of FIG. 3;

FIG. 7 is a flow diagram of the rotary selector valve shown in perspective with a portion thereof broken away;

FIG. 8 is a cylinder diagram of the transformer; and FIG. 9 is an operational diagram of the transformer.

General description

In the drawings I have shown, as an example of one form in which the invention may be embodied, a hydraulic pressure booster comprising, in general (FIG. 3) a main housing A having an input end embodied in a separate end cap B and an output end embodied in a separate end cap C; a free piston positive displacement pump D embodied in a plurality of pump units (sets of relatively large diameter drive pistons and relatively small diameter pumping pistons operable in cylinders of correspondingly differential diameter); a rotary control valve E; a valve driving rotary hydraulic motor F; and a flow control valve G (FIG. 4).

Input drive is provided for by fuel under pressure admitted to inlet cap B through a fuel inlet port 10. As directed by the control valve E, the compressed fuel flows sequentially to driving pistons 11 of the free pistons pump D, the drive pistons 11 being thereby driven forwardly against spring loading and transmitting the forward movement to respective pump pistons 12, which likewise are spring-loaded. At a later point in a cycle of operation, a set of pistons that has been thus driven forwardly is disconnected from the inlet 10 by rotational advance of the control valve E, a connection is established to a fuel outlet port 13 directly opposite the end of control valve E, and this set of pistons are permitted to return in response to their spring-loading, driving the excess fuel through the valve E to the outlet 13.

Valve drive motor F is likewise driven by pressurized fuel which is transmitted from the fuel inlet 10 through a feed passage 14 (FIG. 4) to the leftward end of hydraulic motor F as viewed in FIG. 3. The fuel thus spent in driving motor F likewise finds its way back to the outlet 13 by a route which will be described hereinafter. The motor F is coupled to the valve E for transmitting rotational drive thereto.

Output of high pressure fluid is delivered by the respective pumping pistons 12 of the free piston pump D to a high pressure fluid discharge port 15 in the center of output cap C. The hydraulic fluid of the load circuit is admitted to the pump D through an intake port 16 and a manifold 17 which encircles the cap C.

Detailed description

Main housing section A is fabricated from a suitable corrosion resistant material such as stainless steel. It comprises generally a cup-shaped casting including a generally cylindrical lateral body portion 20 and a bulkhead 21 at its end adjoining the output cap C. Bracket ears 22, for mounting the transformer to a suitable support, are formed integrally with respective ends of lateral body 20 and project tangentially from one side thereof (FIG. 1). Coupling ears 23 and 24, formed at the adjoined ends of housing body 20 and input cap B are connected by cap screws 25 to join the cap B to that end of the body section A. Lateral body 20 has at this input end an integral pilot neck 26 projecting into the cap B and accurately centering the latter in coaxial relation to housing section A. At its output end, housing section A

has a series of tapped bolt holes, equally spaced circumferentially in a concentric series around the major axis of the transformer and receiving the threaded ends of a series of cap screws 27 extending through output cap C. In the outer side of bulkhead 21 is a central recess 28 which accommodates the heads of a pair of cap screws 29 extended therethrough and securing an end unit of the hydraulic motor F.

In the lateral body 20 of housing section A are a series of pump cylinder bores 31 for the pump pistons 12 and a corresponding series of aligned counterbored cylinders 32 for the drive pistons 11. Each cylinder 32 is connected to its respective aligned pump cylinder 31 at an annular radial shoulder 33 which defined the bottom of the respective counterbore 32. The output end of each pump cylinder 31 is connected to a discharge passage 34 extending from the cylinder 31 diagonally in a generally radial inward direction to the adjacent end of main housing section A. The pump cylinders 31 and the discharge passages 34 terminate in a flat end wall 35 disposed in a plane normal to the major axis of the transformer. The pump units, each comprising a pair of aligned cylinders and the pistons therein, are arranged in concentric cylindrical array, equidistantly spaced circumferentially. The outlet ends of each discharge port 34 and its respective pump cylinder 31 are arranged on a common radius of the major axis (FIG. 2). An O-ring seal 36, disposed in an annular groove in the end wall 35 in closely encircling concentric relation to the circular array of outlet ends of discharge passages 34, is sealed to the abutting end face of output cap C, thus directing all of the discharge pressure to the pump outlet port 15 and preventing leakage radially outwardly therefrom.

Main housing section A is provided with a main central cavity comprising respective end bores (valve bore 37 and adapter bore 37') separated by a slightly enlarged intermediate cylindrical chamber 38 housing the hydraulic motor F. For compactness, the bore 37' may have an end portion in the form of an annular, axially projecting recess encircling a crowned central portion of the bulkhead 21.

As shown in FIGS. 1, 2, 5 and 6, the passage 14 for feeding the hydraulic motor F with driving fluid, has an inlet end portion 14' extending diagonally inwardly in a boss 39 from an inlet port which communicates with fuel inlet 10. At its opposite end, passage 14 communicates with a bore which has a counterbore 145 disposed within a diagonally outwardly extending boss 39' mounting the flow valve G. The counterbore 145 is connected to a generally radial passage 14'' extending inwardly to the adapter bore 37' of the housing.

Input cap B comprises an annular skirt portion 40 defining a toroidal inlet chamber 41, a collar portion 42 of reduced diameter joined to the skirt portion 40 by an annular shoulder 43, and a further reduced internally threaded neck portion 44 defining the fuel outlet port 13 and joined to the collar portion 42 by a radial shoulder portion which has a flat radial internal shoulder 45. The shoulder 45 constitutes the bottom of a cylindrical counterbore 46 in the collar portion 42 and in the skirt portion 40. The pilot neck 26 of main body section A is snugly received in the counterbore 46 and is sealed thereto by an O-ring seal 47 disposed in an annular groove in skirt portion 40 and encircling the pilot neck 26. The mounting ears 24 are integral with skirt portion 40. An internally threaded inlet boss 48 defining the inlet 10, is integrally joined to skirt portion 40 on a diagonal axis. The inlet 10 is connected to the inlet chamber 41 by a fuel inlet port 49.

The input cap B (and likewise the output cap C) is preferably fabricated of a light weight material such as aluminum. The outer portion of counterbore 46 is lined with a corrosion resistant bearing bushing 50 in which the rotary selector valve E has its outer end rotatably jour-

nalled. The inner end of valve E is rotatably mounted in the valve bore 37 which has the same diameter as the inner wall of bushing 50 and is axially aligned therewith.

A flat thrust bearing washer 51, seated on shoulder 45, abuts the outer end of valve E.

Output cap C is disclosed in FIGS. 1 and 3 and the adjoining end of main housing section A is seen in FIG. 2.

Output cap C comprises a relatively thick circular collar portion 55 having a main counterbore 56 and a larger counterbore 57. The latter is defined by the cylindrical inner wall of the manifold 17 which consists of a circular annulus of tubular section interrupted by an integral internally threaded boss 58 which defined the pump inlet port 16, communicating with manifold 17.

Pump outlet port 15 is defined by an internally threaded annular cylindrical adapter 59 having a cylindrical outer wall fitted in the counterbore 56 and having at its outer end a radial flange 60 which is received in the outer counterbore 57. The cap screws 27 are extended through apertures in the flange 60 and through registering bores in the collar portion 55 and are threaded into registering bolt holes in the main housing section A, thus securing the adapter 59 within the collar 55 while simultaneously securing the latter to the housing section A. The flange 60 seats against a flat radial shoulder defining the bottom of counterbore 57 and the outer end of collar 55. Adapter 59 is sealed to end cap collar 55 by an O-ring 61 disposed in an annular peripheral groove in fitting 59 and sealed against the wall of counterbore 56.

In the adapter 59 are a series of axially extending bore holes 62 which are axially aligned with the outlet ends of respective discharge passages 34 of main housing section A. A cup shaped discharge check valve poppet 63 is axially slidable in each bore 62, its open end facing the closed bottom thereof and spring-loaded by a coil spring 64 under compression between said closed bottom and the head of the poppet. Said poppet head has an axially projecting valve lip 65 adapted to seat against the adjacent flat radial end face of the main housing section A around the outlet end of a respective discharge passage 34 so as to close the latter except when hydraulic fluid is being forced therethrough by advancing movement of a respective pump piston 12. The bore holes 62 are vented to the pump outlet 15 by respective radial breathing ports 66 extending through the inner wall of adapter 59.

The ends of pump cylinders 31 are open in the flat radial end wall 35 of main housing section A and in communication with axially aligned counterbores (intake valve bores) 67 in the collar portion 55 of end cap C. In the bottom of each intake valve bore 67 is inset a valve seat washer 68 which frames a check valve inlet port 69. There is an annular series of the inlet ports 69, aligned with the respective bores 67 and cylinders 31 and in communication with the inlet manifold 17. A corresponding series of inlet check valve poppets 70 are mounted for axially sliding movements in the respective inlet valve bores 67 and are spring loaded by coil springs 71 engaged against respective spring seat rings 72 which are disposed partially in the bores 67 and partially in the pump cylinders 31. Poppets 70 are of cup-shape and each is provided with an annular series of apertures 73 surrounding the central portion of its head which is normally seated against its respective washer seat 68. The check valve poppets 70 may be of aluminum or other suitably soft material, with the valve seat inserts 68 being of hardened steel or equivalent material. The outlet check valve poppets 63, on the other hand, may be hardened steel, with their seating lips 65 ground to a close fitting seat against the adjacent end face 35 of the softer housing body A.

At this point it may be noted that an inlet flow path from inlet port 16 is provided through each inlet check valve port 69, the apertures 73 of poppet 70, and past the respective spring seat ring 72 into a respective pump cyl-

inder 31, so that during spring-loaded retracting movement of the respective pump piston 12, hydraulic fluid may be drawn into the pump cylinder 31. Upon the forwardly driven stroke of the pump piston 12, the respective inlet check valve poppet 70 will close against its seat 68, closing the respective inlet valve bore 67, and the trapped hydraulic fluid will be forced through the discharge passage 34, unseating the respective discharge check valve 63 and passing therefrom directly into the discharge port 15 through the center of outlet adapter 59. The breathing ports 66 provide for a breathing action behind the poppets 63 to avoid any hydraulic resistance to the unseating movements of these poppets.

Free piston pump D comprises the respective sets of drive pistons 11 and pump pistons 12, axially slidable in the respective cylinders 32 and 31. Pistons 11 and 12 are of cup-shape, having respective skirts enclosing respective end portions of respective return springs 75 and 76. Spring 75 is engaged under compression between the head of piston 11 and a spring seat washer 77 seated on offset shoulder 33. To determine the limit of its retracting movement, the head of piston 11 engages against a stop finger 78 projecting from the center of a stopper 79 which closes the input end of bore 36, being sealed thereto by an O-ring 80. Stopper 79 seats against the adjoining flat end face of input cap B. Return spring 76 is engaged under compression between the head of pump piston 12 and the spring seat ring 72. The latter may have, in the portion received in cylinder 31, a flange projecting radially beyond the portion that projects into intake valve bore 67, such flange seating against the adjoining end wall of output cap C and being held there by the compressive loading of spring 76 (adapted to overcome the opposing force of spring 71, a weaker spring). Each spring 76 is installed to provide a spring-load which may be approximately 40 pounds, in which case the fuel piston return springs 75 may each be installed to provide a spring load of about 25 pounds. The pistons 11 and 12 may be fabricated from "Graph-Air" heat treated to 60 Rockwell hardness and designed with clearance of .005-.0015 inch in their respective cylinders.

Driving pressure is transmitted from each drive piston 11 to its associated pump piston 12 through a stem 81 which may be of a light weight metal such as aluminum, the end of stem 81 bearing directly against the head of piston 12, through the opening in the center of spring seat ring 77, but without being attached to piston 12. It should be noted that the arrangement described above eliminates the necessity for attaching the pistons to one another or making them of one-piece construction, and, furthermore, it eliminates the requirement for critical bore concentricity and alignment which exists where such attached or integral construction is utilized.

The pump piston 12 is provided with a ring groove which receives an O-ring seal 82 of Teflon (tetrafluoroethylene) or equivalent seal material, thereby preventing any fuel leakage from the cylinder 32 to the connected cylinder 31 at the pump side of the apparatus. This eliminates the necessity for extremely close fitting of piston 12 in cylinder 31, although a piston fitted within a tolerance of .0005 inch will act as a secondary seal.

Between each stopper 79 and the associated drive piston 11 there is defined an annular pressure chamber 85 which communicates through a port 86 in housing body A with the control valve bore 37 in an area that thereof where a passage of valve E may recurrently place the port 86 in communication with the annular inlet chamber 41, so that fuel under pressure may be admitted to chamber 85 to advance the drive piston 11 in its bore 32. At the other end of piston 11, the cylinder 32 communicates through a radial port 87 in the housing body A, with the motor chamber 38, to provide a breathing action behind the piston 11 and to drain off, to the fuel outlet, any fuel that may pass the piston 11 from its pressure chamber 85.

Control valve E comprises a central valve body of cru-

ciform section embodying four divider vanes 90 (FIG. 6) projecting radially from the major axis of the apparatus in respective planes of that axis and intersecting one another at right angles at the axis. Defined between the divider vanes 90 are a pair of diametrically opposed inlet passage 91 (FIG. 3) and an intervening pair of diametrically opposed outlet passages 92 at 90 degrees from the inlet passages 91. The ends of inlet passages 91 are sealed to the valve bore 37 by triangular end sectors 93 and 94 respectively, the sides of which are joined to the ends of the vanes 90, terminating at the faces of the vanes 90 which define the outlet passages 92, whereby the latter are open at both ends. The valve E further includes a cylindrical shell 95 which is integrally joined to the peripheral margins of vanes 90 and end sectors 93, 94 and which projects beyond these parts to provide respective end skirts 95' and 95". The shell 95 is rotatably journaled within the valve bore 37 and bushing 50. The inlet passages 91, at their respective ends, terminate in arcuate ports 96 and 96' in the shell 95. Ports 96 are in constant communication with the annular inlet chamber 41 and ports 96' are positioned to register consecutively with the housing body ports 86, providing inlets to the respective drive pressure chambers 85. The areas of shell 95 extending arcuately between the ports 96 seal off the annular inlet chamber 41 from the outlet passages 92.

The arcuate width of inlet passage 91 is such as to provide inlet pressure to diametrically opposed drive pressure chambers 85 for a sufficient extent of revolution of valve E to provide for the full advancing movement of two diametrically opposed drive pistons 11, with overlapping connections to adjacent drive pressure chambers 85 such that the driving strokes of their respective drive pistons 11 may commence while the stroke of the preceding pair of diametrically opposed pistons are being completed.

Chambers 98 and 98', defined within the rearward skirt 95' and the forward skirt 95" of the valve shell 95, are in communication with the open ends of the return passages 92. Rear valve chamber 98 is in open communication with the motor chamber 38.

While the flow of incoming fuel is being applied to diametrically opposed drive pistons 11, intervening pairs of diametrically opposed drive chambers 85 will be disposed in communication with return passages 92 through a pair of diametrically opposed arcuate return ports 97 in shell 95, the ports 97 being positioned to register with the drive chamber ports 86 of the main housing while communication between the return passages 92 and the inlet chamber 41 is closed off by the imperforate areas of the shell 95 extending from the ports 97 to the forward end of the shell (FIG. 7). It will now be apparent that return flow of fuel from the drive chambers 85 through diametrically opposite ports 86 will enter the registering return passages 92 of the valve and will thence flow into the forward chamber 98' of the valve and thence through the outlet 13, sealed off from the inlet chamber 41 and inlet 10.

The fuel used in driving the motor F has return flow from the motor chamber 38 through the rear valve chamber 98, thence through the return passages 92, the forward valve chamber 98' and the outlet 13.

At its rear end, valve E has an integral, internally splined hub 99 for receiving drive from the motor F.

Hydraulic motor F comprises a cylindrical motor case 105 having at its forward end a radial pilot flange 106 in the form of a plurality of segments interrupted by circumferentially spaced slots 107 which provide the communication between the motor chamber 38 and the rear valve chamber 98. At the same end, shell 105 has a head 108 providing a counterbore in which is mounted an anti-friction bearing 109. A splined shaft 110 is coupled in the splined socket of valve hub 99 and is mounted in the bearing 109 for free rotation in housing 105. At its rear end, shaft 110 has an integral head and projecting stem 111 which projects rearwardly through an inclined thrust

bearing comprising a stationary race 112 and a rotating race 113, both of flat ring form, and an interposed circumferential array of rolling bearing elements 114, positioned by a suitable retainer ring as shown. The stationary race 112 is fixed in a mounting ring 115 anchored against rotation by a suitable anchor pin or pins 116. The rear end of stem 111 is secured to a rotating cylinder block 117 which has an array of bores extending parallel to the major axis of the apparatus and disposed in a circumferential array concentric with said axis. In these bores are respective pistons 118, the forward ends of which bear against the rotating race 113 of the inclined thrust bearing and the rearward ends of which are exposed to the pressure of fuel delivered to open rear ends of the piston bores. The rear end of cylinder block 117 has a suitable trunnion (not shown) supported in a needle bearing 120 which is mounted in a hardened valve plate 121. Plate 121 is secured to the rear end of motor case 105 by a retainer ring 122 threaded onto the rear end of the case, and is sealed to the case by an O-ring 123. Cylinder block 117 has a smooth flat squared rear end face which is sealed to the forward face of valve plate 121 by close fitting contact therewith.

An inlet adapter 125 of thick circular disc form is mounted in the adapter bore 37' of the main central housing cavity and is sealed thereto by O-rings 126, 127. Between these O-rings, and between the periphery of adapter 125 and bore 37' there is defined an annular inlet chamber 128 with which the radial portion 14" of the motor drive fuel feed passage is connected. From annular inlet chamber 128, an elbow passage 129 extends through the adapter 125 to the forward face thereof which bears against the valve plate 121. The outlet end of elbow passage 129 is sealed to the valve plate 121 by an O-ring 130 encircling said outlet end. Valve plate 121 is spring-loaded into end-face contact with adapter 125 by means of a coil-spring 136 engaged under compression between its forward face and a retainer ring 137 of wedge shape which bears against the inclined rear face of bearing mounting ring 115. The spring force also holds the ring 115 in place.

The elbow passage 129 communicates with a kidney port 131 extending arcuately and concentrically around the major axis in one side of valve plate 121 and communicating with several of the piston bores of cylinder block 117. The fuel pressure thus applied to the pistons 118 cause them to react against the freely rotatable, inclined bearing race 113 with an action best described as "rolling down-hill," and in doing so, the pistons transmit rotating movement to the cylinder block 117 which consequently rotates and transmits the rotation through stem 111 and splined shaft 110 to the valve E. At the same time, the flat rear end face of cylinder 117, rotating against the forward face of valve plate 121, sequentially presents the open rear ends of the bores of pistons 118 to the kidney port 131 so that the pressurized fuel supply connections to the piston bores are continually indexed to the trailing bores so as to maintain the rotational movement.

As the leading bore of the group in pressurized communication with kidney port 131 breaks communication with that port, it moves into communication with an exhaust kidney port 132 in valve plate 121, through which the used fuel is exhausted as the leading piston is retracted into the cylinder block 117 by rising up the incline to the high side of the bearing race 113. Exhaust port 132 extends arcuately and concentrically around the major axis approximately 180 degrees from the point where the pistons 118 commence to retract to the point where they are fully retracted. Kidney port 132 is connected through an outlet connection 133 to an exhaust chamber 134 in adapter 125, from which the spent fuel passes through a connecting port 135 into the motor chamber 38. From the motor chamber 38, the fuel returns through the valve E as previously described.

Fuel leakage between the fitted contacting end faces of cylinder block 117 and valve plate 121 escapes from the interior of motor shell 105 through registering ports 138 in retainer ring 137 and shell 105, into the motor chamber 38, where it joins the return flow from discharge chamber 134.

The hydraulic motor F is a conventional unit manufactured by Hydro-Aire Company, a Division of Crane Co., of Burbank, California, and identified as Hydro-Aire Motor Part No. 68-193 utilized as a drive for an air compressor for the F-4 H aircraft. Because it is a known unit, the details of construction of the motor are shown herein only to the extent seen in FIG. 3 and the foregoing description is primarily for the purpose of illustrating its adaptation in the hydraulic transformer of the present invention.

Flow control valve G (FIG. 4) may be of any compact capsule type such as that shown, wherein a cup-shaped valve element 141, axially slidable in a barrel 142, is loaded by a spring 143 so as to seek a position of maximum opening in which its skirt uncovers a port 144 in the side of barrel 142. The valve is mounted in the diagonal counterbore 145 in boss 39', with its forward end communicating with feed passage 14 through an aperture 146 in the head of valve cup 141 and with the lateral port 144 communicating, through an annular chamber surrounding the barrel 142, with the radial feed port 14" leading to the inlet chamber 128 of the motor F. The arrangement is such that an increase in pressure in feed port 14 (tending to increase the flow) will act against the head of valve 141 to shift the valve to a position to restrict the lateral port 144, thus regulating the flow to a selected constant volume per unit of time.

Operation

The operation of my improved hydraulic transformer has been partially described as to separate phases thereof, in the preceding description. Reviewing the same generally, the action of the constant speed rotary fuel inlet valve E is such that twice per revolution of the valve and drive motor F, each set of driving and pumping pistons 11, 12, moving in unison as a pair, will make one hydraulic fluid pumping and return stroke, the pumping stroke in response to pressurized fuel acting on the head of drive piston 11 in its drive chamber 85 and the return stroke being energized by the return springs 75, 76 acting independently upon the drive and pumping pistons of the pair. The differential area within drive bore 32 between the two pistons is connected to the fuel outlet 13 through port 87, so that no fluid is trapped, and hydraulic lock is thus avoided.

Sufficient total main piston stroke is allowed so that the maximum flow admitted in drive chamber 85 during the time the same is opened to the inlet 10 through a respective inlet passage 91 of valve E, will produce a piston stroke falling short of the maximum possible stroke permitted by the length of unobstructed space behind piston 11 in bore 32. Thus it is impossible for the pistons to suffer damage from impact with spring abutment rings 77. An excess of room for stroke of the pumping piston 12 is provided so that it likewise is protected against impact damage. The spring-loaded impact of the pistons against stop finger 78 and abutment ring 77 (acting as a return stop for the small piston 12) is insufficient to damage the pistons.

The fuel leakage from the rotary valve E (e.g. between its skirt 95" and bore 37) is connected directly to the return through the rear end chamber 98 of the valve through return passage 92 to the outlet 13. Similarly, leakage past the drive pistons 11 will return directly through the respective ports 87 to the motor chamber 38 and thence through the valve E to outlet 13. Also, any fuel leakage through the drive motor F (e.g. between valve plate 121 and cylinder block 117) will escape from within the motor shell 105 to motor chamber 38 and

thence to outlet 13 through the valve E. Such losses, although they are energy losses, are relatively small. There is no possibility of fuel leaking into the hydraulic fluid in the output end of the transformer, since the Teflon cap seals on the pumping pistons 12 will prevent such leakage. It is estimated that there is only five p.s.i. pressure differential to cause leakage on the intake stroke, and the hydraulic fluid leakage into the fuel flow on the intake end of the transformer will be so small that it can be considered negligible.

Fuel entering the torroidal inlet chamber 41 through main inlet 10 at 1800 p.s.i. will be distributed from the inlet chamber 41 through the diametrically opposed inlet passages 91 of valve E and the respective ports 96' thereof through respective drive chamber ports 86 into the respective drive chambers 85 of respective diametrically opposed drive cylinders 32, simultaneously advancing the respective drive pistons 11 and their associated pumping pistons 12. Hydraulic fluid in the pumping cylinders 31 will thus be forced through discharge passages 34, unseating the check valves 63 thereof, into the pumping outlet 15. At the same time, drive cylinders 32 and pumping cylinders 31 at 90 degrees to the pumping units thus being actuated on their pumping strokes, will be connected through their respective ports 86 (now functioning as outlet ports) through diametrically opposed ports 97 in valve shell 95, to the respective outlet passages 92 of the valve and thence through these passages and the forward end chamber 98' thereof to the fuel outlet 13.

In the pumping cylinders 31 of the pumping units that are thus exhausting fuel through the return passages of valve E, the pumping pistons 12, moving in their return strokes under the yielding pressure of their return springs 76, will draw fuel into these cylinders 31 through the inlet check valves, 70, the outlet check valves 63 having meanwhile closed upon cessation of outlet pressure thereagainst.

The pumping operations described above will occur automatically in response to the control action of valve E which is being rotated by the drive motor F in response to the pressure of fuel bypassed from inlet port 49 through motor feed passages 14', 14, 14" past flow control valve G which regulates the flow to a selected rate. The pressure fuel thus delivered to the operating fluid inlet chamber 128 of motor F will be delivered through elbow passage 129 to inlet kidney port 131 of valve plate 121 and will act upon the motor pistons 118 to effect the rotation of cylinder block 117 in the manner herein above described. This rotation will be transmitted through shaft 110 to the valve E to rotate the valve at the selected rate of revolution. Fuel exhausted from the bores of cylinder block 117 will be discharged through discharge chamber 134 and port 135 to motor chamber 38 from which it will flow to outlet 13 as previously described.

The differential in diameters of drive cylinders 32 and pumping cylinders 31 is such that 1800 p.s.i. fuel pressure will be multiplied into 4150 p.s.i. output pressure (at zero gallons per minute flow rate) in the pumping outlet 15.

My improved hydraulic transformer functions as a variable displacement hydraulic pump in a unique and efficient manner. The flow rate from the pumping outlet 15 is a function of the differential between the approximately 4000 p.s.i. piston balance pressure and the system pressure (the term "piston balance pressure" as here used, refers to the hydraulic fluid pressure acting on the pumping piston 12 which hydrostatically balances the fuel pressure acting on the associated drive piston 11). The larger this pressure differential, the larger the average piston velocity, due to increased acceleration forces. Because the control valve E is regulated to a constant rate of rotation, each piston has a fixed time for its stroke cycle, and as a result, increasing piston velocities will result in increasing piston strokes and con-

sequently, higher pump flow rates. At pump flow rate approaching zero, the pump stroke approaches zero and the only unit power losses are in the rotary valve leakage and the total rotary valve drive motor flow. Such losses are estimated to be approximately 7 percent of the maximum input power.

Referring now to FIGS. 2 and 6, it will be noted that the invention permits a fairly compact arrangement of pumping cylinders in annular array around the valve E and motor F, with a satisfactory proportioning of diameters as between the drive cylinders and the main central housing cavity, such that ten sets of drive and pumping cylinders may be accommodated in the circumferential extent of the housing. Referring now to FIGS. 8 and 9, the cylinders are numbered therein to illustrate the timing of the pumping strokes of the respective pairs of diametrically opposed pumping units with reference to the strokes of the other units. In each of these schematic figures, the letter "P" indicates pumping flow and the letter "R" indicates return flow.

It may now be noted that cylinders 1 and 6 will be simultaneously actuated in their pumping strokes, while at the same time cylinders 3 and 8 will be starting their return strokes. This is indicated in FIG. 9. FIG. 8 likewise indicates this condition and, in addition, shows that cylinders 2 and 7 will be starting their return stroke while cylinders 1 and 6 are completing theirs; and that cylinders 4 and 9 will be starting their return strokes while cylinders 3 and 8 are completing theirs. Thus, the stroking of the piston sets of the free piston pump is progressive and overlapping in character so that a relatively constant pumping output is provided, with only minor fluctuations therein.

The return of the pistons can be done not only by means of springs located in the piston bores, but also by applying pressure to the back of the pistons during the return stroke. By simply connecting the back side of the driving piston to the top side of a piston located 90° apart, the pressure-return sequence in each piston will be such that both sides of the driving piston will be under it but with opposite sequence creating a positive return stroke and avoiding the springs.

By using the principle described in the preceding paragraph, the pump side could easily be converted into a double-acting pump, meaning the pump could use each stroke to pressurize fluid at one side of the piston and simultaneously absorb fluid into the opposite side, thus doubling the flow capacity maintaining the length of the stroke.

I claim:

1. In a fluid pressure transformer apparatus, in combination: a free-piston pump comprising a plurality of piston cylinder sets each comprising respective drive and pump pistons of differential diameter and cylinders of correspondingly different diameter in which said pistons are respectively slidable, said drive pistons being arranged to transmit pumping strokes to said pump pistons; a drive fluid inlet for said pump; a rotary control valve operable to direct flows of drive fluid of one pressure from said inlet against said drive pistons in their respective cylinders in a cycling sequence, for effecting said pumping strokes; valve means for controlling intake of fluid into said pump piston cylinders and discharge of fluid under another pressure from said pump; and a rotary fluid motor having an inlet port connected to said drive fluid inlet to receive a portion of the inlet flow of said drive fluid and directly driven by said inlet flow, said motor having a driving connection to said valve for rotating the valve to effect cycling of the inlet flows to the cylinders of said drive pistons, said valve and motor being arranged end-to-end in axial alignment on the major axis of the apparatus, and said piston-cylinder sets being arranged in concentric array around said valve and motor, with their respective axes parallel to said major axis.

2. Apparatus as defined in claim 1, including a flow control valve operable on the flow to said motor inlet port to regulate said flow at a selected constant value.

3. Apparatus as defined in claim 1, wherein said valve includes return flow passages for controlling discharge of drive fluid from the cylinders of said drive pistons.

4. In a fluid pressure transformer apparatus, in combination: a housing comprising a generally cylindrical tubular wall structure defining a central cavity on its major axis and having a plurality of pairs of aligned relatively large and small diameter cylinders extending parallel to said major axis within said wall structure, said relatively large cylinders being arranged in a series extending toward one end of said cavity and said relatively small cylinders being arranged in a series extending toward the other end of said cavity; each of said series of cylinders being arranged in circumferentially-spaced concentric array around said central cavity; means defining drive fluid inlet and outlet ports at one end of said housing, for conveying relatively low pressure drive fluid; means defining pumping intake and discharge ports at the other end of the housing; pump pistons in the cylinders of one of said series operable to draw fluid into their cylinders through said intake port and to exhaust it at relatively high pressure through said discharge port; drive pistons in the cylinders of the other series operable to transmit pumping strokes to said pump pistons; a control valve rotatable in said central cavity in said one end of the housing, said valve having an inlet passage operable to direct flows of said low pressure fluid in cycling sequence from said inlet port to the cylinders of said other series for driving said drive pistons to effect said pumping strokes of said pump pistons; and a motor in said central cavity, directly coupled to an end of said valve and operable to rotate the valve to effect said cycling, said motor being a hydraulic motor connected to said inlet port for receiving a portion of the flow therefrom and being directly driven by the inlet flow of drive fluid.

5. In a fluid pressure booster apparatus, in combination: a housing comprising a generally cylindrical tubular wall structure defining a central cavity on its major axis and having a series of pairs of aligned relatively large and small diameter cylinders extending parallel to said major axis within said wall structure, and joined at their inner ends, said relatively large cylinders extending toward one end of said housing and said relatively small cylinders extending toward the other end of the housing; said cylinders being arranged in circumferentially-spaced concentric array around said central cavity; means defining low pressure inlet and outlet ports at said one end of said housing; means defining high pressure hydraulic fluid intake and discharge ports at said other end of the housing; pump pistons in said relatively small diameter cylinders, operable to draw hydraulic fluid thereinto through said intake port and to exhaust said hydraulic fluid through said discharge port; drive pistons in said relatively large cylinders, operable to transmit pumping strokes to said pump pistons; a control valve rotatable in said central cavity in said one end of the housing, said valve having in its periphery an inlet passage operable to direct flows of said low pressure fluid in cycling sequence from said inlet port to said relatively large cylinders for driving said drive pistons to effect said pumping strokes of said pump pistons, and having a return passage circumferentially spaced from said inlet passage, extending axially in said valve and communicating through an end thereof with said low pressure outlet; and a motor in said central cavity, directly coupled to an end of said control valve and operable to rotate the valve to effect said cycling.

6. Apparatus as defined in claim 5, wherein said motor is a hydraulic motor connected to said low pressure inlet port for receiving a portion of the flow therefrom, said motor being directly driven by said low pressure inlet flow.

7. In a fluid pressure booster apparatus, in combina-

tion: a housing comprising a generally cylindrical tubular wall structure defining a central cavity on its major axis and having a series of pairs of aligned relatively large and small diameter cylinders extending parallel to said major axis within said wall structure and joined at their inner ends by an annular offset shoulder, said relatively large cylinders extending toward one end of said housing and said relatively small cylinders extending toward the other end of the housing; said cylinders being arranged in circumferentially-spaced concentric array around said central cavity; an input cap detachably secured to said one end of the housing, said input cap defining low pressure inlet and outlet ports for circulation of low pressure drive fluid; an output cap detachably secured to said other end of the housing, said output cap defining high pressure hydraulic fluid intake and discharge ports; pump pistons in said relatively small diameter cylinders, operable to draw hydraulic fluid thereinto through said intake port and to exhaust said hydraulic fluid through said discharge port at relatively high pressure in a circulation path isolated from that of said drive fluid; drive pistons in said relatively large cylinders, operable to transmit pumping strokes to said pump pistons; a control valve rotatable in said central cavity in said one end of the housing, said valve having an inlet passage operable to direct flows of said low pressure fluid in cycling sequence from said inlet port to said relatively large cylinders for driving said drive pistons to effect said pumping strokes of said pump pistons, and having a return passage circumferentially spaced from said inlet passage, extending axially in said valve and communicating through an end thereof with said low pressure outlet; and a motor in said central cavity, directly coupled to an end of said control valve and operable to rotate the valve to effect said cycling.

8. In a fluid pressure intensifier apparatus, in combination: a housing comprising an inner wall structure defining a cylindrical central cavity on its major axis and a radially outward body portion having a series of pairs of aligned relatively large and small diameter cylinders extending parallel to said major axis within said outward body portion and joined at their inner ends, said relatively large cylinders extending toward one end of said housing and said relatively small cylinders extending toward the other end of the housing; said cylinders being arranged in circumferentially-spaced concentric array around said central cavity; means defining low pressure inlet and outlet ports at said one end of said housing for circulating a drive fluid; means defining high pressure hydraulic fluid intake and discharge ports at said other end of the housing; pump pistons in said relatively small diameter cylinders, operable to draw hydraulic fluid therein through said intake port and to exhaust said hydraulic fluid through said discharge port at relatively high pressure in a circulation path isolated from that of said drive fluid; drive pistons in said relatively large cylinders, operable to transmit pumping strokes to said pump pistons; said housing having in said inner wall structure a plurality of radial ports communicating with respective large diameter cylinders and with said central cavity, and having a toroidal inlet passage axially spaced from said radial ports toward the end of the housing, communicating with said central cavity and with said low pressure inlet port; a cylindrical control valve rotatable in said central cavity, said valve having in its periphery an inlet passage operable to successively connect said radial ports to said toroidal passage so as to direct flows of said low pressure fluid in cycling sequence from said inlet port to said relatively large cylinders for driving pistons to effect said pumping strokes of said pump pistons, and having a return passage circumferentially spaced from said inlet passage and having one end thereof communicating through the periphery of said valve with respective radial ports in sequence, said return passage extending axially in said valve and communicating through the outer end thereof with said outlet port

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for return flow of low pressure fluid from said large diameter cylinders; and means for transmitting rotary drive to said valve.

9. Apparatus as defined in claim 8, wherein said rotary drive transmitting means comprises an hydraulic motor 5 disposed in said central cavity inwardly of said valve, directly coupled thereto, receiving a portion of the flow of said low pressure fluid and directly driven thereby.

10. Apparatus as defined in claim 9, including means 10 for regulating the rate of flow of said low pressure fluid to said hydraulic motor so as to regulate the speed of operation of said intensifier apparatus.

11. Apparatus as defined in claim 8, wherein said low pressure outlet port is disposed in the end of the housing on said major axis and said low pressure inlet port is 15 disposed radially outwardly of said outlet port.

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References Cited by the Examiner

UNITED STATES PATENTS

2,243,978	6/1941	Reader	-----	103—49
2,762,307	9/1956	Orloff	-----	103—49
2,897,762	8/1959	Houvenier	-----	103—49
3,068,794	12/1962	Morris et al.	-----	103—44 X
3,077,838	2/1963	Maglott	-----	103—45
3,085,512	4/1963	Alyanak et al.	-----	103—49

FOREIGN PATENTS

987,120 3/1949 France.

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